taking place in a majority of slots on both poles (both sides of the rotor), the binding forces would cancel themselves out, building up axial strain forces within the rotor.

If one or more of the windings on predominately one side of the rotor broke free, allowing the rotor to bow instantly, it would cause the vibration step change. The amount of step change would depend on how much thermal strain was in the copper, which would be a function of unit watt and var load (field temperature), the number of windings released suddenly, and their angular location. In addition, the direction of the bow could either add to or subtract from any residual imbalance already present in the rotor, causing either upward or downward movements in the overall vibration amplitude. Adding to the unpredictable nature of the step changes was the fact that the more the unit bound and released, the more the materials binding the windings would wear. In theory, this would both dust and lubricate the windings, and increase the sliding clearances, thus improving the situation.

In September 1990, the plant took the unit off line for an inspection and overhaul. The generator rotor was returned to the repair shop that did the original rewinding and carefully disassembled. As suspected, it was found that the G-11 type insulation (creepage blocking) beneath the slot wedges had been improperly sized. Varying sizes of blocking had been installed during reinstallation of the slot wedges.

Figure 5 illustrates, in an exaggerated form, how the copper bars deformed because of centrifugal force and nonuniform blocking, preventing them from expanding axially. The final repair resolution involved slightly milling the underside of the slot wedges in order to use a uniform strip of creepage block. This created a smooth slip plane between the copper and the blocking insulation, allowing uniform axial expansion of the copper bars.

The unit was back in service 26 November 1990, and has not experienced any further vibration step changes.

Machinery Messages

Case History

The importance of monitoring Shaft Centerline data

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This article discusses the basic theory behind Shaft Centerline monitoring and the problems experienced in a recent field study where startup assistance was provided on main process machinery. A case history is also included which focuses on severe misalignment problems experienced during initial machinery startup. The problem was further compounded by a loose bearing retaining setscrew. This case history demonstrates that Shaft Centerline information is an important part of the diagnostic process as it provides additional information on complex machinery malfunctions.

Introduction

1X and/or 2X Polar and/or Bode plots contribute to being the most useful machinery diagnostic data. Both plots show a machine's amplitude and phase response versus speed. In practice, this type of data presentation is widespread and has become accepted as an industry standard as the "best" plots to observe for machinery diagnostics. However, another data presentation type, Shaft Centerline, is generally overlooked.

Many machinery diagnosticians have long recognized the value of shaft radial position information. The existence of a preload (external or internal) can often be quickly identified by observing the average radial position of the Shaft Centerline within the bearing clearance. Shaft Centerline position may also change as a result of bearing babbitt deterioration due to electrostatic discharge or simply as a result of bearing wear resulting over time.

Monitoring Shaft Centerline data in today's rotating machinery applications can provide important and relevant information to changing machinery conditions.

This concept is illustrated in the following case history in which Shaft Centerline data became the key component in solving a machinery malfunction. But first, a brief discussion is in order to clarify some theoretical aspects regarding Shaft Centerline monitoring and diagnostics.

Shaft Centerline theory

When a noncontacting eddy current proximity transducer and Proximitor® are used to monitor lateral shaft motion, the transducer system provides the following signal components:

1. An AC signal (in this case, negatively fluctuating) which provides shaft dynamic motion relative to the probe mounting.

2. A DC signal which provides the average radial shaft position relative to the probe mounting.

Typically, the dynamic signal is monitored by a radial vibration monitor and is displayed as the amount of overall machine vibration in mils or micrometres peak-to-peak. However, the DC component of this signal is, for the most part, unused. Therefore, it is relevant to note that, just as an axial position probe monitors axial movement, so can a radial proximity probe be used to measure radial shaft position within the bearing.

A typical arrangement for Shaft Centerline monitoring is to have two orthogonal (XY) proximity probes per ►

bearing which provide the required DC signals. Comparing the combination of these signals to a known radial bearing clearance provides sufficient information to determine Shaft Centerline position within the bearing clearance.

In order to obtain *accurate* Shaft Centerline data, an initial zero speed gap voltage reference is required. This reference is generally obtained with the rotor at rest or on turning gear. In this condition, it is assumed that the rotor is at rest in the bottom of its bearings; therefore, all subsequent gap voltage measurements are referenced to this initial starting position. As machine speed increases during startup, measurement of changing gap voltages from the transducers indicates the amount of average shaft travel within that bearing. At operating speed, the rotor's average position within the bearing is then easily identified.

Analyzing this position within the known diametral bearing clearance provides valuable information on a number of important parameters:

1. Coupling and bearing alignment
2. Rotor preloading
3. Oil film thickness
4. Bearing wear.
5. The attitude angle, α.

Attitude angle is classically defined as the angle between the direction of the steady state load on the rotor (such as gravity or fluidic forces) whose direction may be unknown and a line connecting the geometric bearing centerpoint and the centerline of the shaft.

The attitude angle can be obtained from Shaft Centerline data quite easily as long as care has been taken in determining the correct bearing clearance and zero speed gap voltage reference. In general, evaluation of the shaft attitude angle provides an indication of the stability margin for a particular machine. As long as machine alignment is correct, and the parameters for Shaft Centerline measurements have been observed, an attitude angle approaching 90° is indicative of instability within the system. Refer to Figure 1 (normal) and Figure 2 (abnormal).

As a general rule, an ideal rotor position in the bearing for a clockwise machine would be in the lower left-hand quadrant with an attitude angle, α, between 20 and 60 degrees, as measured from the vertical.

Similarly, for a counter-clockwise machine, the rotor position would be in the lower right-hand quadrant with the same attitude angle being measured from the vertical. The convention of observation is always taken looking from the driver (motor/turbine) to the driven (generator/compressor).

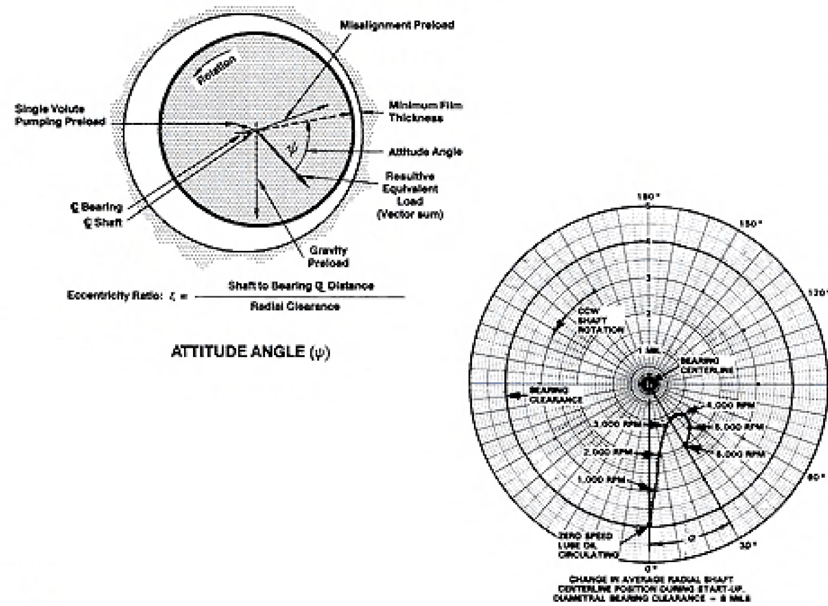

Figure 1

Shaft attitude angle definition (left). An example of how the shaft radial average centerline position changes during startup (right).

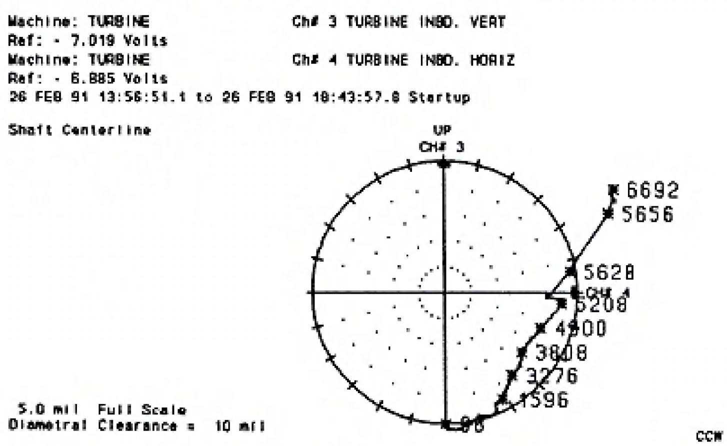

Figure 2

Shaft centerline data showing excessive rotor travel beyond the diametral bearing clearance at 6692 rpm. This plot also shows the total rotor travel from startup where α is approximately 130° from vertical.

Case history

The machine train in question consists of an Elliott SEPG-6 six-stage steam turbine, driving an Elliott 38-M8-7 seven-stage centrifugal compressor via a Kopflex gear coupling. Typical operating speed is in the region of 7100 rpm. The process is propane which is used as a refrigerant in the oil refining cycle.

During a major plant turnaround, this particular machine was completely disassembled and overhauled as required. All journal bearings were replaced, the compressor rotor reworked and the turbine rotor rebuilt. Upon reassembly, machine alignment was performed by plant personnel using pre-outage hot and cold alignment data.

Initial 1X Bode and Polar plots showed little residual imbalance present in the system. In fact, the highest vibration amplitude was slightly over 2 mils (51 μm) peak-to-peak through the first balance resonance. By all indications, the machine appeared to operate normally.

Further inspection of orbital and spectral data did not indicate a malfunction. As time progressed, however, steady state data on both turbine bearings showed sudden minor changes in amplitude and phase although no changes were observed at the compressor bearings. Only by looking at Shaft Centerline data was it apparent that something was terribly wrong. As shown in Figure 2, the inboard turbine bearing shaft position exceeded the diametral clearance by 2 mils (51 μm) at operating speed.

According to plant records, alignment of this unit was performed with pre-turnaround hot and cold readings using Essinger Bars, in conjunction with reverse dial indicators. Initial thoughts regarding the malfunction focused upon a wiped bearing at the exhaust end of the turbine (Bearing #2). The machine was monitored constantly for changes, and over certain compressor loading ranges, changes were experienced. These changes came in the form of increasing radial vibration at both turbine bearings which resulted in the machine tripping off-line on several occasions.

Violent compressor surges during the outset of the project were believed to play a large part in these sudden amplitude and phase changes. Analysis showed that, due to changing preloads on the turbine rotor at the time the compressor was surging, alignment was now questionable.

As plant personnel were making provisions for a forced shutdown to check alignment, examination of OEM drawings, specifically of the inboard turbine bearing, drew suspicions toward a bearing retaining setscrew. The assumption was, if this screw were loose, the effect of having no upper bearing restraint would be seen, which would be responsible for rotor travel beyond the diametral bearing clearance.

This setscrew is located on the top of the inboard bearing carrier (Figure 3) and is the sole component for restraining the upper bearing carrier. The recommended OEM torque setting for this setscrew is 20 lb ft (27 N•m). The screw is accessible upon removal of an external coverplate on the bearing housing. Subsequent inspection (prior to shutdown) revealed this screw was loose!

Plant management decided to tighten the screw with the machine running at normal operating speed. Figure 4 clearly shows the resulting abrupt change in Shaft Centerline position which forced the rotor to return to a position within the diametral bearing clearance as the screw was tightened.

This action increased the rotor preload significantly, which was reflected mainly in the orbital data along with a shift in 1X amplitude and phase at both turbine bearings. For the most part, the compressor was unaffected.

When the machine was finally shut down, alignment between the turbine and compressor was found to be incorrect, primarily in the vertical plane.►

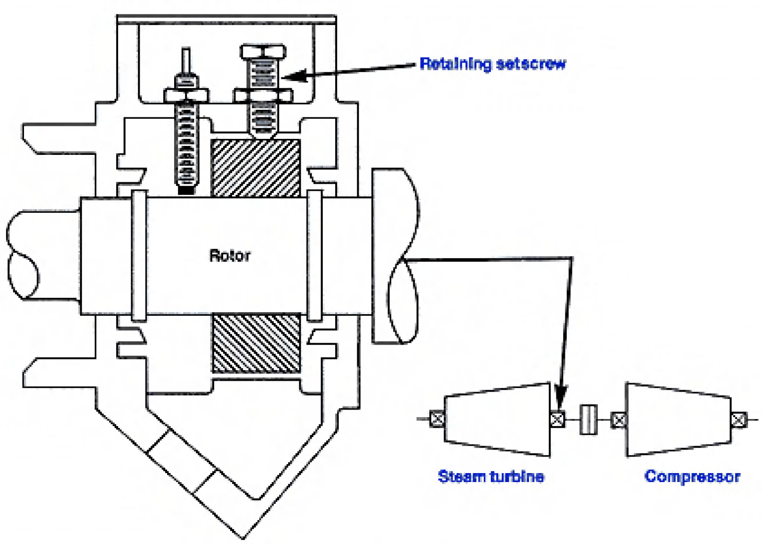

Figure 3

Machine drawing showing the inboard turbine bearing. Note the location of the bearing retaining setscrew.

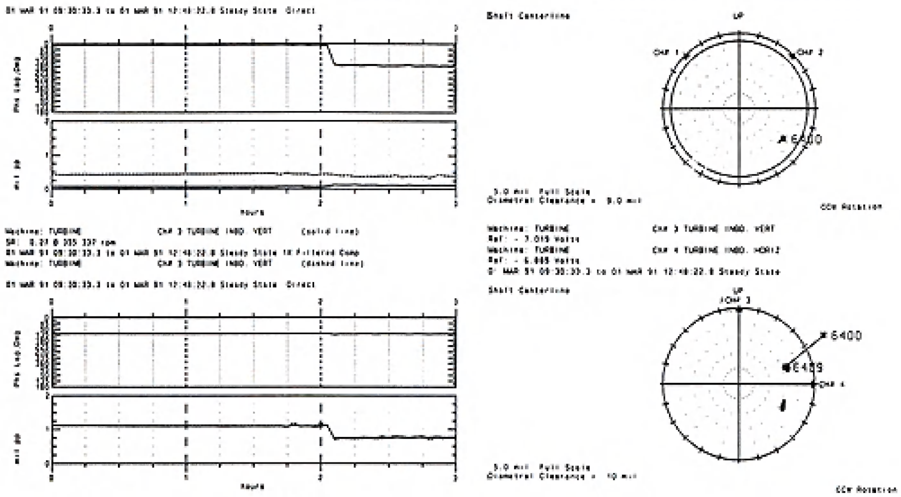

Figure 4

Changes in 1X amplitude and phase (left), along with the change in Shaft Centerline data (right) occurred when the inboard bearing setscrew was torqued to 20 lbs ft (27 N•m). This action resulted in increased rotor preload and confirmed that misalignment was present in the system.

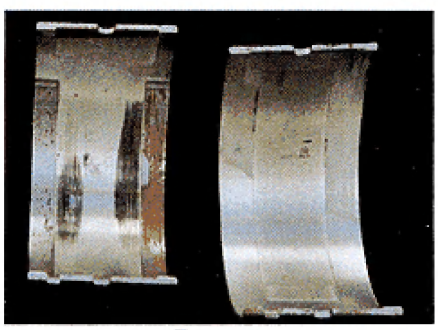

Figure 5

Inboard turbine bearings showing wear due to misalignment and the loose setscrew.

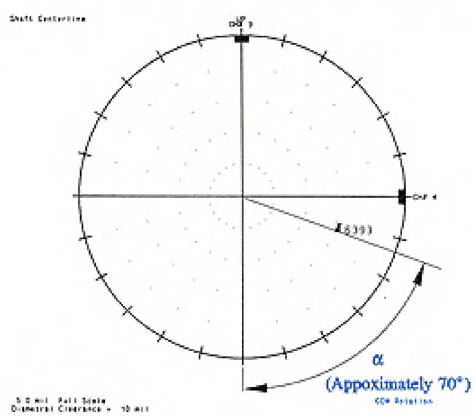

Figure 6

The final running speed position at the inboard turbine bearing. Note the attitude angle, α, has been reduced from 130° to approximately 70°.

By looking at the lower Shaft Centerline plot (Figure 4), it became obvious that the rotor position was still too high at the inboard bearing location. The turbine alignment figures were recalculated based on the latest set of hot readings, and thermal growth factors were calculated from the last cold readings. Actual corrective alignment moves on the turbine, incorporating Shaft Centerline data as an additional reference, were:

Horizontal: The turbine case was moved to the right 0.003 inch (79 μm). This resulted in the rotor being moved toward the center of the plot.

Vertical: The turbine case was raised 0.009 inch (229 μm). This resulted in the rotor being moved toward the lower half of the bearing.

Remember, for a counterclockwise, horizontally-mounted machine, the ideal rotor position would be in the lower right-hand quadrant with α between 20 and 60 degrees.

A subsequent bearing inspection found damage to both upper and lower bearing shells on the turbine inboard side. The upper bearing shell exhibited wear in the same location as the rotor position shown in Figures 2 and 4. Refer to Figure 5. Both shells were replaced and the bearing clearance was checked using Plastigage. The diametral clearance was recorded at 0.010 inch (254 μm).

By using Shaft Centerline data, in conjunction with regular alignment techniques, the realignment process was performed with a ***higher*** degree of confidence. Figure 6 shows the final operating speed position of the rotor within the bearing. Since this procedure, the machine has not experienced any further problems.

Conclusion

By monitoring Shaft Centerline data, it was possible to focus directly on the malfunction and to obtain a faster remedy than by using 1X Bode and Polar plots alone. Without the advantage of Shaft Centerline data, the problem may have continued undetected and eventually caused unnecessary downtime and consequential cost increases.